# FASTENER PRE-STRESSING JOINT

## CROSS-REFERENCE TO RELATED APPLICATION

**[0001]** This claims the benefit of US Provisional Patent Application No. 60/516,488 filed October 31, 2003.

## STATEMENT CONCERNING FEDERALLY SPONSORED RESEARCH OR DEVELOPMENT

**[0002]** Not applicable.

## FIELD OF THE INVENTION

**[0003]** This invention relates to fastened joint design, such as a bolted joint, and in particular to a joint that pre-stresses a fastener to result in a more uniform stress in the fastener at maximum application load.

## BACKGROUND OF THE INVENTION

**[0004]** Bending stress is induced across the shank 11 of a fastener such as a bolt in the plane of bending when the joint is non-symmetric and when the fastener loading is not on the centerline of the fastener. Figs. 1A and 1B schematically illustrate two fasteners with the same axial application load (i.e., the load at the axis of the shank of the bolt). Fig. 1A illustrates a bending stress superimposed in the plane of the paper (in the plane of bending, so as to make the fastener shank 11 convex left) on the axial pre-stress so as to subject the shank 11 of the fastener 10 to bending and axial loads. Fig. 1B illustrates the shank 11 of the fastener 10 subjected to only an axial load (no bending load), equal in magnitude to the axial load 12 at the center axis of the bolt in Fig. 1A, so that the average stress in both fasteners is the same.

[0005] The bending stress in the fastener in Fig. 1A reduces the load carrying capacity of the fastener and joint. One side 14 of the fastener shank 11 in the plane of bending has higher stress than the other side 16 because of the induced bending. This is not a desirable condition because the stress distribution across the fastener shank causes high stress on side 14 of the fastener shank. A more desirable stress condition at maximum loading would be to have a uniform stress distribution across the fastener shank 11 in the plane of bending at maximum loading conditions as illustrated in Fig. 1B, where the stresses 12, 14 and 16 are substantially equal. In some cases, fastened joints cannot be designed to eliminate bending stresses in the fastener under all conditions, such as in a connecting rod joint where the application load is dynamic and therefore changes.

[0006] The load being carried by the fastener is related to the average stress in the fastener. In Figs. 1A and 1B, both fasteners 10 have the same average stress 12 but the fastener in Fig. 1A has a higher maximum stress 14, as a result of the bending stress. If failure occurs it would occur at the point of highest stress along side 14. Thus, bending stress added to the axial stress reduces the load carrying capability of a fastener compared to a fastener subjected to the same average stress but with a uniform stress distribution.

[0007] Referring to Fig. 2A, when fastening a joint, an initial axial pre-stress is applied as a result of tightening or tensioning the fastener. This is represented by the uniform pre-stress components 18. If the joint is non-symmetric, it will compress more on one side than the other side of the fastener hole. This causes the fastener shank 11 to be subjected to bending stresses and the load to be applied in a non-uniform fashion across the fastener shank 11. This is represented by the non-uniform components 20. In addition, if the application load is applied off-center to the fastener centerline, additional

bending will occur in the fastener. The stress components 12, 14 and 16 in Fig. 1A are the sum of the uniform components 18 and the non-uniform components 20 at maximum application load. Fastener joint design is limited by the highest stress level in the fastener including the bending stress, which makes a uniform stress profile as illustrated in Fig. 1B a more desirable choice.

**[0008]** The word "fastener" as used herein is any type of fastener having a shank that is subjected to tensile forces when applied to a joint, such as bolts, rivets, rods (threaded, pinned, welded, etc.), screws, etc. The word "bending stress" refers to a non-uniform stress across the fastener shank. This invention includes the use of a nut in the joint system, which could act like a bolt head and therefore "head" of a fastener includes a nut, a bolt head or screw head, a rivet head or rivet flange, etc.

## SUMMARY OF THE INVENTION

**[0009]** This invention provides a bolt joint that at the maximum loading conditions in the service application of the joint the maximum stress will be reduced across the bolt shank. It does this by the joint inducing a bending stress in the fastener shank in the plane of bending of the application bending stress when the fastener is assembled to the joint. The bending stress induced by the joint is substantially inversely proportional to the bending stress induced in the plane of bending by the maximum application load that the fastener shank is subjected to in service so as to reduce the maximum stress when the maximum application load is applied.

**[0010]** By so doing, the invention also reduces the cyclical mean stress to which the fastener shank is subjected. This is especially useful to increase the fatigue life of the fastener.

[0011] In a useful aspect of the invention, the bending stress induced by the joint is of a magnitude and direction to produce a substantially uniform stress distribution across the fastener shank in the plane of bending when the maximum application load is applied, to obtain the full advantage of the invention.

[0012] In one form of the invention, the joint has a seat that the fastener bears against to induce tension in the shank and the seat is skewed at an angle other than 90 degrees to an axis of a fastener hole in the parts through which the shank extends. The seat is angled in a direction so as to induce bending stresses in the fastener opposite in direction to the bending stresses induced by the maximum application load. Thereby, the bending stresses induced by the joint cancel the bending stresses induced by the application load to reduce the maximum application load on the shank of the fastener and to reduce the cyclic mean stress to which the fastener shank is subjected.

[0013] In another way of practicing the invention, the joint has joint faces that face one another and are held together by the fastener, a portion of the joint faces defining between them an unsupported gap that induces bending stresses in the shank of the fastener opposite in direction to bending stresses induced by the maximum application load.

[0014] In another ways of practicing the invention, a hole that extends in the parts and receives the fastener shank has a first portion in one of the parts and a second portion in the other part, with the first portion skewed relative to the second portion so as to induce bending stresses in the fastener opposite in direction to bending stresses induced by the maximum application load.

**[0015]** These different ways of practicing the invention can be practiced alone or in any combination with one another.

**[0016]** In an especially useful form, the joint is a joint in a connecting rod connecting a bearing cap to a rod portion of the connecting rod. A bearing cap joint is an especially useful application of the invention because the fastener shank is subjected to a cyclic bending stress by the cyclic motion of the connecting rod, such that prestressing the fasteners using the invention can reduce the maximum application stress and the cyclic mean stress in the fastener shanks.

**[0017]** The foregoing and other objects and advantages of the invention will appear in the detailed description which follows. In the description, reference is made to the accompanying drawings which illustrate a preferred embodiment of the invention.

## BRIEF DESCRIPTION OF THE DRAWINGS

**[0018]** Fig. 1A is a typical prior art bolt stress distribution diagram illustrating the stress distribution in a bolt subjected to bending and axial loading;

**[0019]** Fig. 1B is a typical prior art bolt stress distribution diagram illustrating the stress distribution in a bolt subjected to only axial loading, with the magnitude of axial loading equal to the loading at the axis of the bolt in Fig. 1A;

**[0020]** Fig. 2A is a view like Fig. 1A with the stress diagram illustrating the components of total stress as pre-stress and maximum stress;

**[0021]** Fig. 2B is a view of a fastener comparable to Fig. 2A, but with a pre-stress and maximum stress distribution produced by a joint incorporating the invention;

**[0022]** Fig. 3 is a view of a connecting rod bearing cap joint with an angled bolt seat according to the invention, the angle being exaggerated for illustrative purposes;

[0023] Fig. 4 is a view like Fig. 3 but of a typical prior art connecting rod bearing cap joint;

[0024] Fig. 5 is a view of a connecting rod bearing cap joint with angled joint faces according to the invention, the angles being exaggerated for illustrative purposes; and

[0025] Fig. 6 is a view of a connecting rod bearing cap joint with the threaded fastener holes in the rod being angled inwardly according to the invention, the angles being exaggerated for illustrative purposes.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

[0026] Referring to Fig. 2B, the present invention provides a fastener clamped joint design that provides a substantially uniform stress load distribution in the fastener shank 11 at the maximum application loading conditions. In Fig. 2B, the stress chart shows the initial pre-stress with a lower set of vector arrows 22, 24 and 26 and the application stress by the upper set of vector arrows 28, 30 and 32. In both Fig. 2A and Fig. 2B the average pre-stress and the average maximum stress are the same so each case would be handling the same system load; however using the invention results in a lower maximum stress under the application load. In the case where the joint application loads are cyclic as in a connecting rod bearing cap joint, the mean cyclic stress would be lower also. The total cyclic stress swing would remain the same.

[0027] The schematic stress charts of Figs. 2A and 2B are simplified in that they do not show any incidental or accidental joint bending pre-stress. If there was joint bending pre-stress, the horizontal set of pre-stress vectors in Fig. 2A would be non-

uniform (at some angle) and the corresponding pre-stress vectors in Fig. 2B would need to be adjusted to compensate for the bending pre-stress.

[0028] Uniform stress distribution at maximum application loading can be accomplished in any number of ways. Currently, typical connecting rod bearing cap joints are made as illustrated in Fig. 3, with each bolt joint seat 36 oriented 90° to the corresponding bolt hole 37 and threaded hole 39 centerline 38, the unthreaded hole 37 being in the bearing cap 42 and the threaded hole 39 being in the connecting rod body 44. This yields a stress distribution substantially as in Fig. 2A, with the vectors 18 representing the static pre-stress and the vectors 20 representing the dynamic application loading. Note that in this case, the maximum stress occurs on the inner side (toward the crankshaft bore 40 of both seats 36.

[0029] One way to practice the present invention would be to skew each joint bolt seat 36 to the bolt hole 37 and threaded hole 39 centerline 38 by some small amount, chosen based on the maximum application loading that is to be cancelled or offset. Typically, the angle would be less than one degree, for example .125 degrees, depending on the magnitude of application loading. The angle must also be in the correct direction so that it cancels the bending stress at the maximum application (dynamic) loading condition, which is induced by the joint and application load. This is illustrated in Fig. 3. Both seats 36, which are flat as illustrated, are machined or formed so as to both angle or skew inwardly in the direction of the plane of bending, so as to induce bending stresses in each bolt 10 that are counter to the bending stresses induced by the application load. In other words, the bolts 10 tend to bow outwardly (convex-out relative to the axis of the main bore 40) in the plane of the paper as a result of the skewed seats 36, whereas the

application load tends to bow the bolts 10 inwardly (convex-in relative to the axis of the main bore 40). The magnitude and direction of the angle of the seats 36 is chosen, and also the torque to which the bolts 10 are tightened is chosen, so as to produce a substantially uniform stress distribution in the shank of the fastener 10 at the maximum application load, as illustrated in Fig. 2B.

**[0030]** If in Fig. 4 the bolt hole 37, 39 and bolt-joint seat are machined along the same spindle centerline 38, the seat and bolt centerline would be 90 degrees to each other by virtue of the manufacturing process, like the typical joint shown in Fig. 3. An additional or different process is needed to create the required bolt seat 36 skewness. This could be done in many different ways. For example, the bolt seat 36 skewness of Fig. 4 could be forged into the bearing cap 42. Another way would be to machine the bolt hole with one spindle along axis 38 and machine the bolt seat with another spindle at a small angle to the hole-drilling spindle. Yet another way would be to create the angle of the seats 36 by using the powder metallurgy process to form the skewness of each bolt seat 36 in the bearing cap 42.

**[0031]** Another way to create a uniform stress across the bolt shank 11 in the plane of bending at maximum application load is to make the joint faces, where they face each other near the center of the main bore 40, at a small angle to each other tapering outwardly so as to create a small unsupported gap 48 between each set of the joint faces in the area adjacent to the bore 40. This is illustrated in Fig. 5. One or both facing surfaces could be angled so as to create the gap 48. This small angle (greatly exaggerated in Fig. 5; may be less than one degree depending on the magnitude of the application load to be cancelled) could be machined on the faces, formed by forging or powder

metallurgy, or the joint could be plastically deformed to create the gap, which latter method could be incorporated into an otherwise typical fracture splitting production process of a rod and cap of a connecting rod. This allows the cap 42 to flex toward the rod member 44 in the areas of the gaps formed by the angles, which has the effect of subjecting the shanks 11 of the fasteners 10 to bending stresses so as to bow them outwardly. When the bolts 10 are tightened, the gap 48 may be closed or substantially closed, or not. The size of the gaps 48 and the torque to which the bolts 10 are tightened are chosen so as to produce a substantially uniform stress distribution in the shank 11 of the fastener 10 in the plane of bending at the maximum application load, as illustrated in Fig. 2B.

**[0032]** Yet another way to create a uniform stress in the plane of bending across the bolt shank 11 at maximum application load would be to create the centerline 38A of the threaded hole 39 at a small angle to the bolt hole 37 and (unbent) bolt 10 centerline 38B as illustrated in Fig. 6. Again, the angles of the axes 38A are greatly exaggerated and may be less than one degree relative to the axes 38B. These bow the shanks 11 of the bolts 10 outwardly, as in the previously described embodiments, to yield a uniform stress distribution across the bolt shank in the bending plane at maximum application load, with a reduced cyclical mean stress and reduced maximum stress in the bolt shank 11. The angles of the axes 38A and the torque to which the bolts 10 are tightened are chosen so as to produce a substantially uniform stress distribution in the bending plane in the shank 11 of the fastener 10 at the maximum application load, as illustrated in Fig. 2B.

**[0033]** Preferred embodiments of the invention have been described in considerable detail. Many modifications and variations to the preferred embodiments

described will be apparent to a person of ordinary skill in the art. Therefore, the invention should not be limited to the embodiments described.